F Dimensions of Steel Pipe

TABLE F.1 Schedule 40

Nominal Pipe Size (in)	Outside Diameter (in)	Outside Diameter (mm)	Wall Thickness (in)	Wall Thickness (mm)	Inside Diameter (in)	Inside Diameter (ft)	Inside Diameter (mm)	Flow Area (ft²)	Flow Area (m²)
1/8	0.405	10.3	0.068	1.73	0.269	0.0224	6.8	0.000 394	3.660×10^{-5}
1/4	0.540	13.7	0.088	2.24	0.364	0.0303	9.2	0.000 723	6.717×10^{-5}
3/8	0.675	17.1	0.091	2.31	0.493	0.0411	12.5	0.001 33	1.236×10^{-4}
1/2	0.840	21.3	0.109	2.77	0.622	0.0518	15.8	0.002 11	1.960×10^{-4}
3/4	1.050	26.7	0.113	2.87	0.824	0.0687	20.9	0.003 70	3.437×10^{-4}
1	1.315	33.4	0.133	3.38	1.049	0.0874	26.6	0.006 00	5.574×10^{-4}
1 1/4	1.660	42.2	0.140	3.56	1.380	0.1150	35.1	0.010 39	9.653×10^{-4}
1 1/2	1.900	48.3	0.145	3.68	1.610	0.1342	40.9	0.014 14	1.314×10^{-3}
2	2.375	60.3	0.154	3.91	2.067	0.1723	52.5	0.023 33	2.168×10^{-3}
2 1/2	2.875	73.0	0.203	5.16	2.469	0.2058	62.7	0.033 26	3.090×10^{-3}
3	3.500	88.9	0.216	5.49	3.068	0.2557	77.9	0.051 32	4.768×10^{-3}
3 1/2	4.000	101.6	0.226	5.74	3.548	0.2957	90.1	0.068 68	6.381×10^{-3}
4	4.500	114.3	0.237	6.02	4.026	0.3355	102.3	0.088 40	8.213×10^{-3}
5	5.563	141.3	0.258	6.55	5.047	0.4206	128.2	0.139 0	1.291×10^{-2}
6	6.625	168.3	0.280	7.11	6.065	0.5054	154.1	0.200 6	1.864×10^{-2}
8	8.625	219.1	0.322	8.18	7.981	0.6651	202.7	0.347 2	3.226×10^{-2}
10	10.750	273.1	0.365	9.27	10.020	0.8350	254.5	0.547 9	5.090×10^{-2}
12	12.750	323.9	0.406	10.31	11.938	0.9948	303.2	0.777 1	7.219×10^{-2}
14	14.000	355.6	0.437	11.10	13.126	1.094	333.4	0.939 6	8.729×10^{-2}
16	16.000	406.4	0.500	12.70	15.000	1.250	381.0	1.227	0.1140
18	18.000	457.2	0.562	14.27	16.876	1.406	428.7	1.553	0.1443
20	20.000	508.0	0.593	15.06	18.814	1.568	477.9	1.931	0.1794
24	24.000	609.6	0.687	17.45	22.626	1.886	574.7	2.792	0.2594

Schedule (Thickness) of Steel Pipe Used in Obtaining Resistance Of Valves and Fittings of Various Pressure Classes by Test*

Valve or Fitting ANSI Pressure Classification – Steam Rating		Cold Rating	Schedule No. of Pipe Thickness
250-Pound and Lower		500 psig	Schedule 40
300-Pound to 600-Pound		1440 psig	Schedule 80
900-Pound		2160 psig	Schedule 120
1500-Pound		3600 psig	Schedule 160
2500-Pound	½ to 6"	6000 psig	xx (Double Extra Strong)
	8" and larger	3600 psig	Schedule 160

*These schedule numbers have been arbitrarily selected only for the purpose of identifying the various pressure classes of valves and fittings with specific pipe dimensions for the interpretation of flow test data; they should not be construed as a recommendation for installation purposes.

Representative Equivalent Length‡ in Pipe Diameters (L/D) Of Various Valves and Fittings

Description of Product				Equivalent Length In Pipe Diameters (L/D)
Globe Valves	Stem Perpendicular to Run	With no obstruction in flat, bevel, or plug type seat	Fully open	340
		With wing or pin guided disc	Fully open	450
	Y-Pattern	(No obstruction in flat, bevel, or plug type seat) – With stem 60 degrees from run of pipe line	Fully open	175
		– With stem 45 degrees from run of pipe line	Fully open	145
Angle Valves		With no obstruction in flat, bevel, or plug type seat	Fully open	145
		With wing or pin guided disc	Fully open	200
Gate Valves	Wedge, Disc, Double Disc, or Plug Disc		Fully open	13
			Three-quarters open	35
			One-half open	160
			One-quarter open	900
	Pulp Stock		Fully open	17
			Three-quarters open	50
			One-half open	260
			One-quarter open	1200
Conduit Pipe Line Gate, Ball, and Plug Valves			Fully open	3**
Check Valves	Conventional Swing		0.5†...Fully open	135
	Clearway Swing		0.5†...Fully open	50
	Globe Lift or Stop; Stem Perpendicular to Run or Y-Pattern		2.0†...Fully open	Same as Globe
	Angle Lift or Stop		2.0†...Fully open	Same as Angle
	In-Line Ball		2.5 vertical and 0.25 horizontal†...Fully open	150
Foot Valves with Strainer		With poppet lift-type disc	0.3†...Fully open	420
		With leather-hinged disc	0.4†...Fully open	75
Butterfly Valves (8-inch and larger)			Fully open	40
Cocks	Straight-Through	Rectangular plug port area equal to 100% of pipe area	Fully open	18
	Three-Way	Rectangular plug port area equal to 80% of pipe area (fully open)	Flow straight through	44
			Flow through branch	140
Fittings	90 Degree Standard Elbow			30
	45 Degree Standard Elbow			16
	90 Degree Long Radius Elbow			20
	90 Degree Street Elbow			50
	45 Degree Street Elbow			26
	Square Corner Elbow			57
	Standard Tee	With flow through run		20
		With flow through branch		60
	Close Pattern Return Bend			50
Pipe	90 Degree Pipe Bends			See Page A-27
	Miter Bends			See Page A-27
	Sudden Enlargements and Contractions			See Page A-26
	Entrance and Exit Losses			See Page A-26

**Exact equivalent length is equal to the length between flange faces or welding ends.

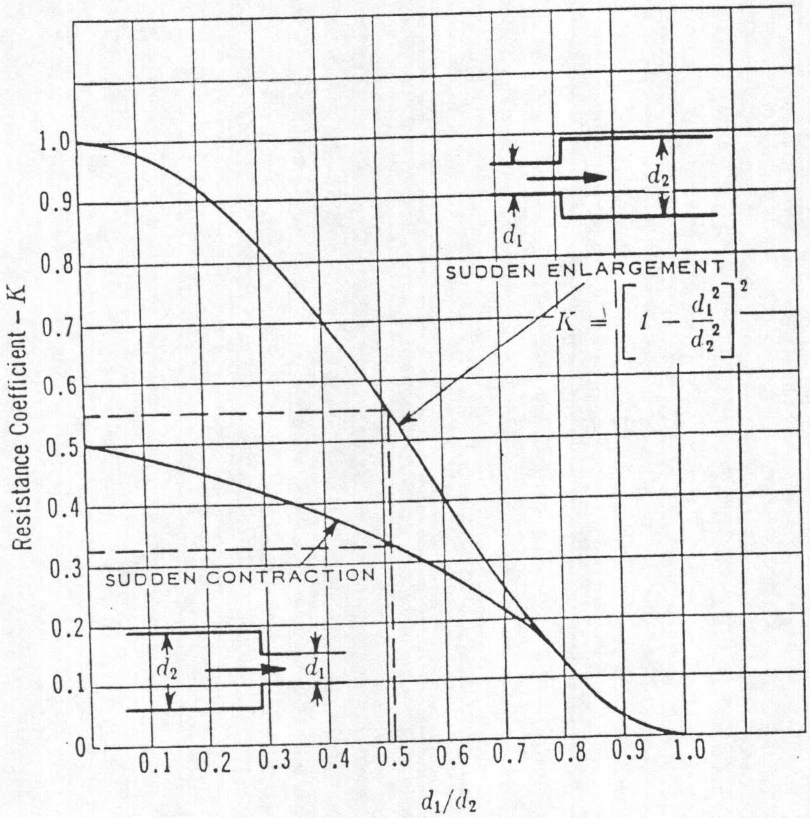
†Minimum calculated pressure drop (psi) across valve to provide sufficient flow to lift disc fully.

‡For limitations, see page 2-11. For effect of end connections, see page 2-10.

For resistance factor "K", equivalent length in feet of pipe, and equivalent flow coefficient "C_v", see pages A-31 and A-32.

Resistance in Pipe

Resistance Due to Sudden Enlargements and Contractions[20]



Sudden enlargement: The resistance coefficient K for a sudden enlargement from 6-inch Schedule 40 pipe to 12-inch Schedule 40 pipe is 0.55, based on the 6-inch pipe size.

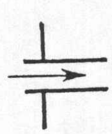
$$\frac{d_1}{d_2} = \frac{6.065}{11.938} = 0.51$$

Sudden contraction: The resistance coefficient K for a sudden contraction from 12-inch Schedule 40 pipe to 6-inch Schedule 40 pipe is 0.33, based on the 6-inch pipe size.

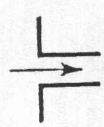
$$\frac{d_1}{d_2} = \frac{6.065}{11.938} = 0.51$$

Note: The values for the resistance coefficient, K, are based on velocity in the small pipe. To determine K values in terms of the greater diameter, multiply the chart values by $(d_2/d_1)^4$.

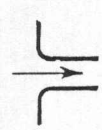
Resistance Due to Pipe Entrance and Exit



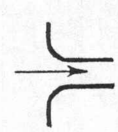
$K = 0.78$
Inward Projecting Pipe Entrance



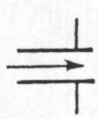
$K = 0.50$
Sharp Edged Entrance



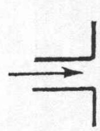
$K = 0.23$
Slightly Rounded Entrance



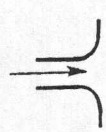
$K = 0.04$
Well Rounded Entrance



$K = 1.0$
Projecting Pipe Exit



$K = 1.0$
Sharp Edged Exit



$K = 1.0$
Rounded Exit

Problem: Determine the total re sistance coefficient for a pipe or diameter long having a sharp edge entrance and a sharp edged exit.

Solution: The resistance of pip one diameter long is small and can b neglected ($K = f\ L/D$).

From the diagrams, note:

Resistance for a sharp edged entrance	= 0.5
Resistance for a sharp edged exit	= 1.0

Then,

the total resistance, K, for the pipe	= 1.5

Resistance of Bends

Resistance of 90 Degree Bends[21]

The chart at the right shows the resistance of 90 degree bends to the flow of fluids in terms of equivalent lengths of straight pipe.

Resistance of bends greater than 90 degrees is found using the formula:

$$\frac{L}{D} = R_t + (n-1)\left(R_l + \frac{R_b}{2}\right)$$

n = total number of 90° bends in coil
R_t = total resistance due to one 90° bend, in L/D
R_l = resistance due to length of one 90° bend, in L/D
R_b = bend resistance due to one 90° bend, in L/D

Problem: Determine the equivalent lengths in pipe diameters of a 90 degree bend and a 270 degree bend having a relative radius of 12.

Solution: Referring to the "Total Resistance" curve, the equivalent length for a 90 degree bend is 34.5 pipe diameters.

The equivalent length of a 270 degree bend is:

$$L/D = 34.5 + (3-1)\,[18.7 + (15.8 \div 2)]$$
$$L/D = 87.7 \text{ pipe diameters}$$

Note: This loss is less than the sum of losses through three 90 degree bends separated by tangents. For "resistance of bends theory", see page 2-12.

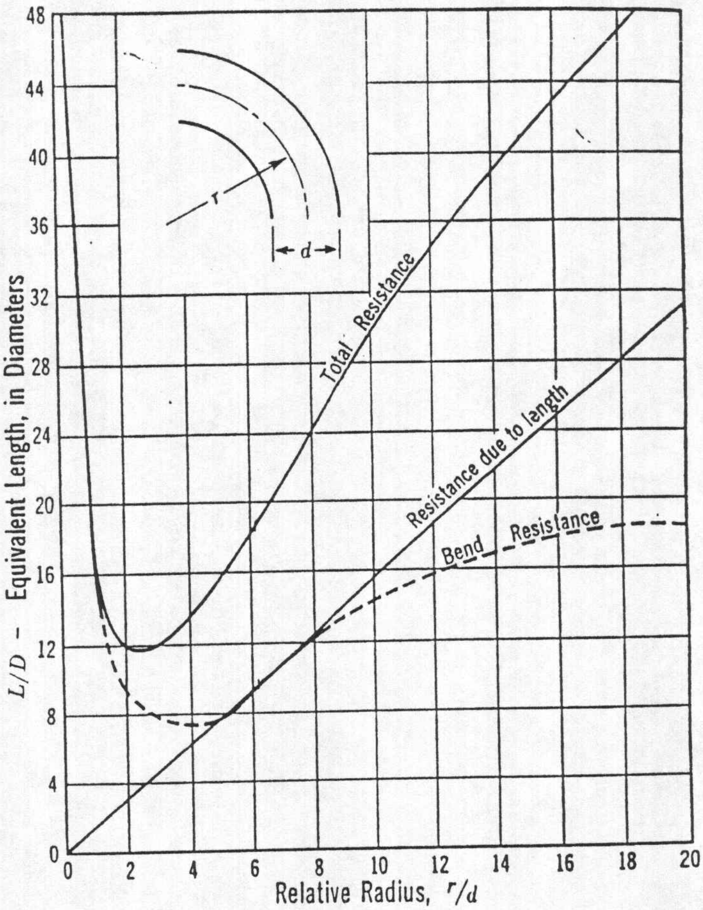


Chart for Resistance of 90 Degree Bends

From *Pressure Losses for Fluid Flow in 90 Degree Pipe Bends* by K. H. Beij. Courtesy of Journal of Research of National Bureau of Standards, Vol. 21, July, 1938.

Resistance of Miter Bends[4]

The chart at the lower right shows the resistance of miter bends to the flow of fluids. The chart is based on data published by the American Society of Mechanical Engineers (ASME).

Problem: Determine the equivalent length in pipe diameters of a 40 degree miter bend.

Solution: Referring to the "Total Resistance" curve in the chart, the equivalent length is 12 pipe diameters.

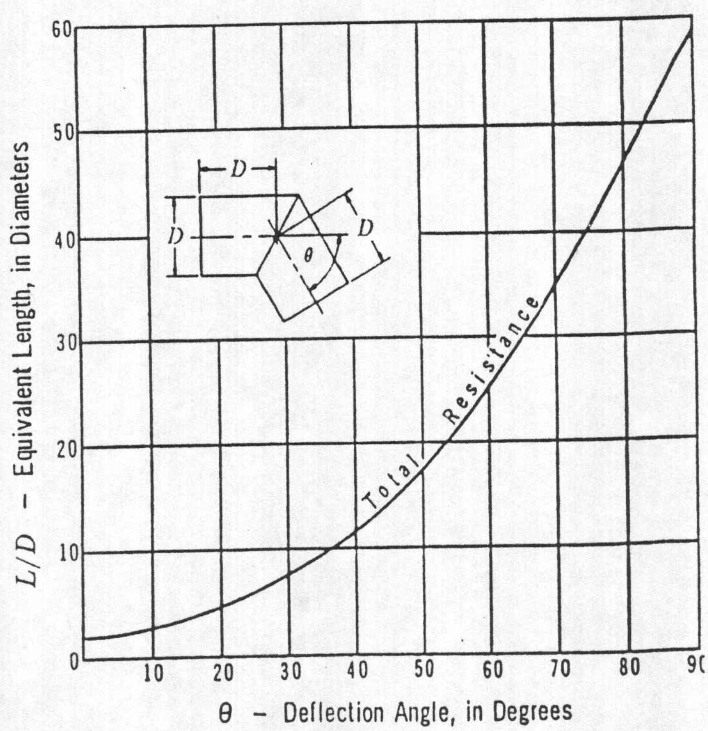
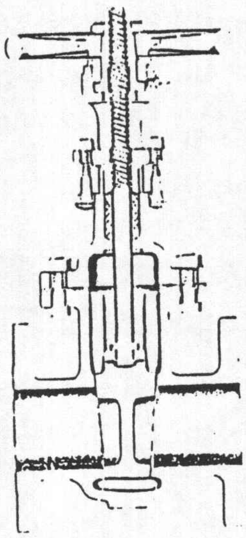


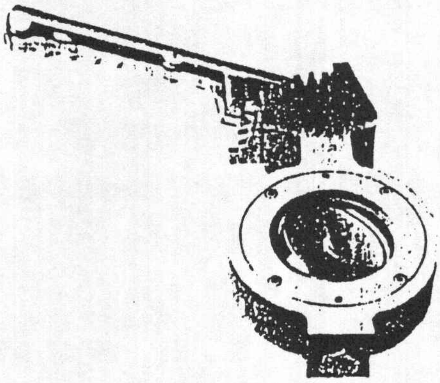
Chart for Resistance of Miter Bends

Types of Valves

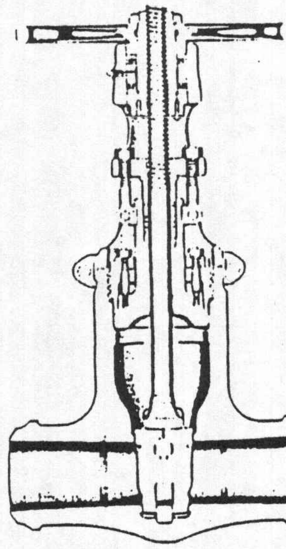
(For other valve types, see page 3-26)



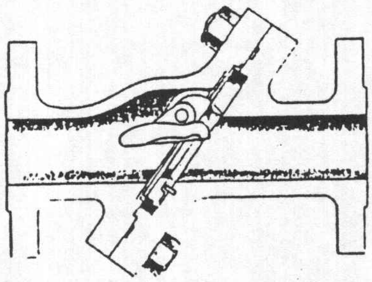
Wedge Gate Valve
(Bolted Bonnet)



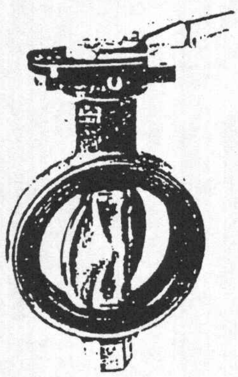
High Performance Butterfly Valve



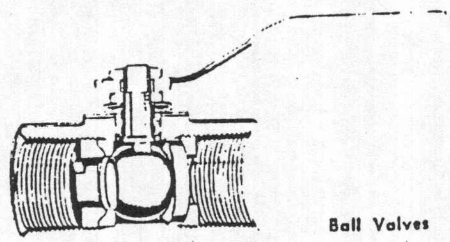
Flexible Wedge Gate Valve
(Pressure-Seal Bonnet)

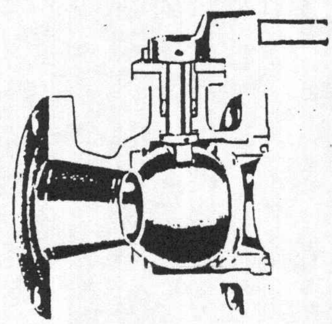


Tilting Disc Check Valve

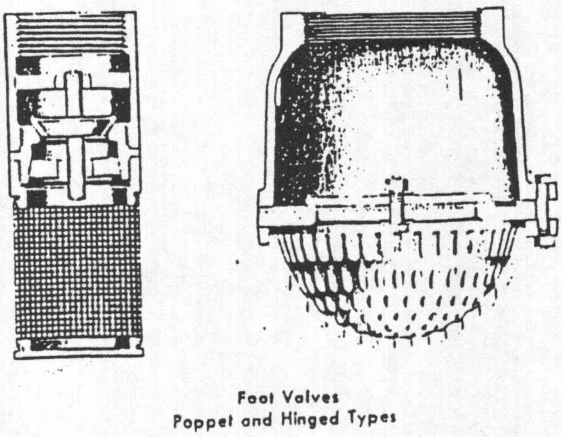


Butterfly Wafer Valve

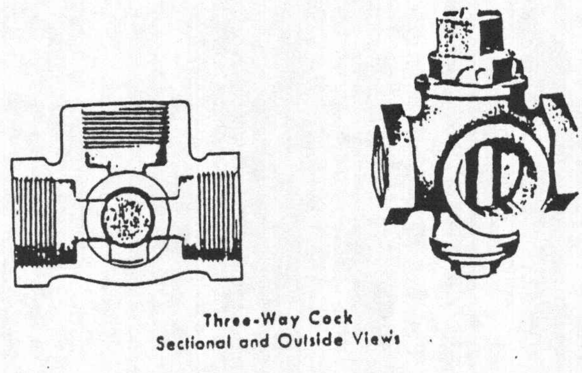




Ball Valves



Foot Valves
Poppet and Hinged Types



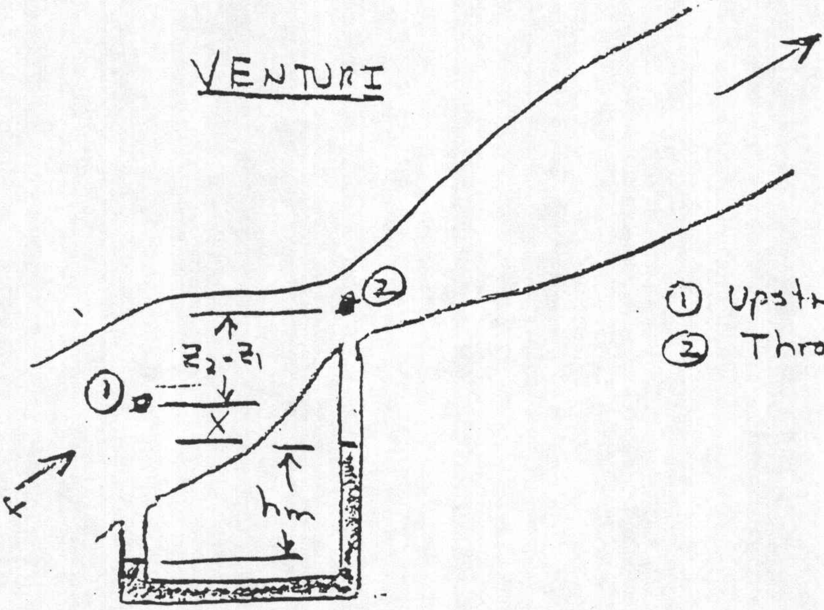
Three-Way Cock
Sectional and Outside Views

Derivation of Flowmeter Equation

Venturi, Nozzle, Orifice

Schedule Run
Venturi Chart

VENTURI



① Upstream
② Throat

Energy Equation

$$\frac{P_1}{\gamma} + \frac{V_1^2}{2g} + Z_1 = \frac{P_2}{\gamma} + \frac{V_2^2}{2g} + Z_2 + h_L$$

$$\frac{V_2^2 - V_1^2}{2g} = \frac{P_1 - P_2}{\gamma} + Z_1 - Z_2 - h_L$$

but $A_1V_1 = A_2V_2$ or $V_1 = \frac{A_2}{A_1}V_2 = \left(\frac{D_2}{D_1}\right)^2 V_2$

then

$$\frac{V_2^2 - \left(\frac{D_2}{D_1}\right)^4 V_2^2}{2g} = \frac{P_1 - P_2}{\gamma} + Z_1 - Z_2 - h_L$$

$$V_2 = \sqrt{\frac{2g\left(\frac{P_1-P_2}{\gamma} + Z_1 - Z_2 - h_L\right)}{1 - \left(\frac{D_2}{D_1}\right)^4}}$$

Manometer Equation

$$P_2 + \gamma_f(Z_2 - Z_1 + \cancel{X}) + \gamma_m h_m - (h_m + \cancel{X})\gamma_f = P_1$$

$$P_1 - P_2 = \gamma_f(Z_2 - Z_1) + h_m(\gamma_m - \gamma_f)$$

OR

$$\frac{P_1 - P_2}{\gamma} = Z_2 - Z_1 + h_m\left(\frac{\gamma_m}{\gamma_f} - 1\right)$$

OR

$$\frac{P_1 - P_2}{\gamma} + Z_1 - Z_2 = h_m\left(\frac{\gamma_m}{\gamma_f} - 1\right)$$

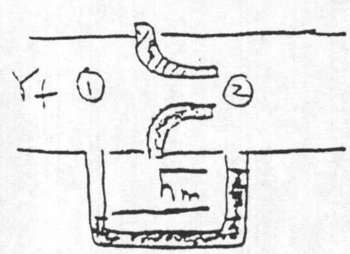
h_L = friction loss, a number greater than 0 that will always reduce the value of V_2 in the equation. h_L would be found from experiment. If we replace h_L with a flow coefficient, C_V, at the front end of the equation, it would have a value of 1 when $h_L = 0$ and would decrease toward 0 as h_L would increase. Its value would also be found from experimentation. Also if we multiply V_2 by A_2 then we have the flowrate, Q.

$$Q = A_2 V_2 = C_V A_2 \sqrt{\frac{2g\left(\frac{P_1-P_2}{\gamma} + Z_1 - Z_2\right)}{1 - (D_2/D_1)^4}}$$, then substituting the manometer equation

$$Q = C_V A_2 \sqrt{\frac{2g \cdot h_m\left(\frac{\gamma_m}{\gamma_f} - 1\right)}{1 - (D_2/D_1)^4}}$$

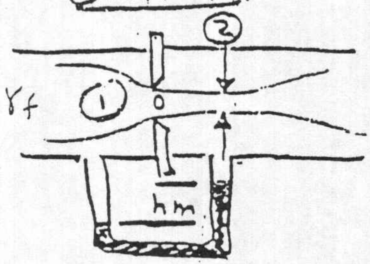
C_V is a function of Reynolds Number, N_R
(See Figure 14.3, pg 436 of the text.)

NOZZLE



Assume $K \approx 1$
or
Crane

ORIFICE



Assume $K =$ 0.6 - 0.62

$A_2 = C_c A_0$
or
Crane

$$Q = C_N A_2 \sqrt{\frac{2g\,h_m\left(\frac{\gamma_m}{\gamma_f}-1\right)}{1-\left(\frac{D_2}{D_1}\right)^4}}$$

OR

$$Q = K_N A_2 \sqrt{2g\,h_m\left(\frac{\gamma_m}{\gamma_f}-1\right)}$$

where $K_N = \dfrac{C_N}{\sqrt{1-\left(\frac{D_2}{D_1}\right)^4}}$

C_N = function of N_R
(Fig. 14.5, pg 438)

K_N = function of N_R, D_2/D_1
(Handout)

$$Q = C_O A_O \sqrt{\frac{2g\,h_m\left(\frac{\gamma_m}{\gamma_f}-1\right)}{1-C_c^2\left(\frac{D_O}{D_1}\right)^4}}$$

OR

$$Q = K_O A_O \sqrt{2g\,h_m\left(\frac{\gamma_m}{\gamma_f}-1\right)}$$

Where $K_O = \dfrac{C_O}{\sqrt{1-C_c^2\left(\frac{D_O}{D_1}\right)^4}}$

C_O = function of N_R, D_O/D_1
(Fig. 14.7, pg 440)

K_N = function of N_R, D_O/D_1 (Handout)

To solve for Q in any of the flowmeters, a single iteration process is necessary, since C_V, C_N, C_O and K_N, K_O are functions of N_R and thus, Q. The following steps should be taken. We will use C + K for general solution.

Step I = Plug all known values into the equation to yield Q = C (Number) OR Q = K (Number)

Step II - Assume $N_R = 10^5$ and select C or K from appropriate graph.

Step III - Substitute C or K into equation from step I, and calculate Q_{old}

Step III - Calculate N_R in upstream pipe, using Q_{old} $N_R = \dfrac{4\rho Q}{\pi d_1 \mu} = \dfrac{4Q}{\pi d_1 \nu}$

Step IV Select new C or N value from appropriate graph.

Single Iteration is enough

Step V Substitute C or K into equation and calculate Q_{New}

(Iteration not required if you know Q and are calculating h_m.)
(Several iterations needed if you know Q, h_m and are calculating d_2.)

But $v_2^2 = v_1^2(A_1/A_2)^2$. Then we have

$$v_1^2[1 - (A_1/A_2)^2] = 2g[(p_1 - p_2)/\gamma + (z_1 - z_2) - h_L]$$

$$v_1 = \sqrt{\frac{2g[(p_1 - p_2)/\gamma + (z_1 - z_2) - h_L]}{1 - (A_1/A_2)^2}} \tag{15–3}$$

We can make two simplifications at this time. First, the elevation difference $(z_1 - z_2)$ is very small, even if the meter is installed vertically. Therefore, this term is neglected. Second, the term h_L is the energy loss from the fluid as it flows from section 1 to section 2. The value of h_L must be determined experimentally. But it is more convenient to modify Eq. (15–3) by dropping h_L and introducing a discharge coefficient C:

$$v_1 = C\sqrt{\frac{2g(p_1 - p_2)/\gamma}{1 - (A_1/A_2)^2}} \tag{15–4}$$

Equation (15–4) can be used to calculate the velocity of flow in the throat of the meter. Note that the velocity depends on the difference in the pressure head between points 1 and 2. This is the reason these meters are called variable head meters.

Normally we want to calculate the volume flow rate.

Since $Q = A_1 v_1$, we have

$$Q = CA_1\sqrt{\frac{2g(p_1 - p_2)/\gamma}{1 - (A_1/A_2)^2}} \tag{15–5}$$

The discharge coefficient C represents the ratio of the actual velocity through the venturi to the ideal velocity for a venturi with no energy loss at all. Therefore, the value of C will always be less than 1.0. The venturi of the Herschel type shown in Figure 15.2 is designed to minimize energy losses by employing a smooth gradual contraction to the throat and a smooth, long enlargement following the throat. Therefore, the discharge coefficient is typically close to 1.0.

Figure 15.3 indicates that the actual value of C depends on the Reynolds number for the flow in the main pipe. Above a Reynolds number of 2×10^5 the value of C is taken to be 0.984. This value applies to a venturi of the Herschel type that

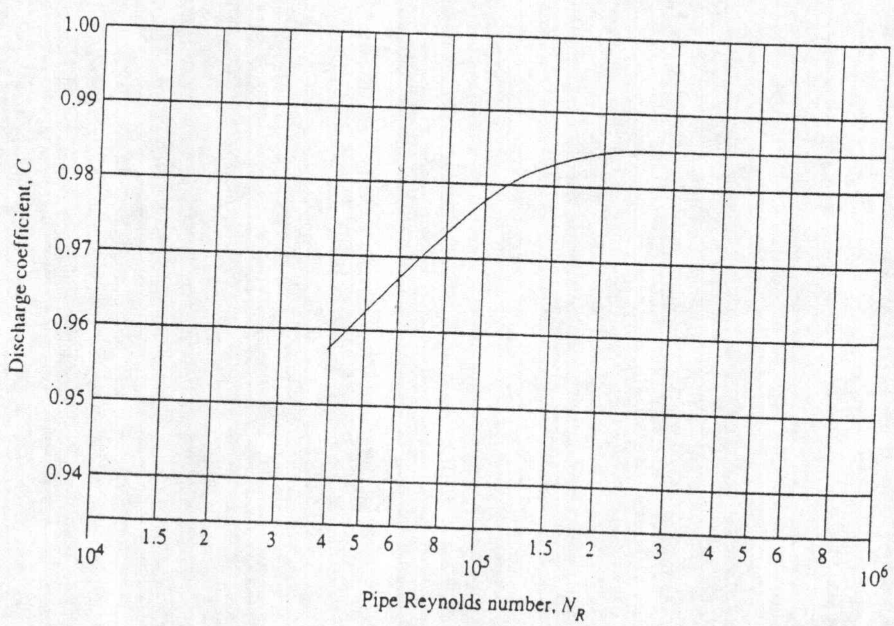
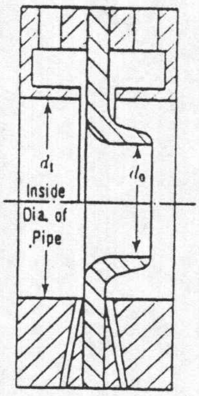


FIGURE 15.3 Discharge coefficient for a rough cast venturi tube of the Herschel type. (Source: ASME Research Committee on Fluid Meters. 1959. *Fluid Meters: Their Theory and Application*. 5th ed. New York: The American Society of Mechanical Engineers, p. 125.)

Flow Coefficient C for Nozzles[7]

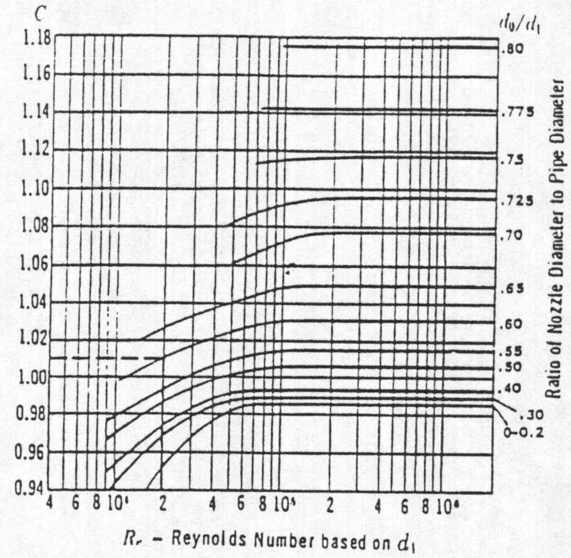


Flow →

Data from *Regeln fuer die Durchflussmessung mit genormten Duesen und Blenden*. VDI-Verlag G. m.b. H., Berlin, SNW, 7, 1937. Published as Technical Memorandum 952 by the NACA.

$$C = \frac{C_d}{\sqrt{1 - \left(\frac{d_0}{d_1}\right)^4}}$$

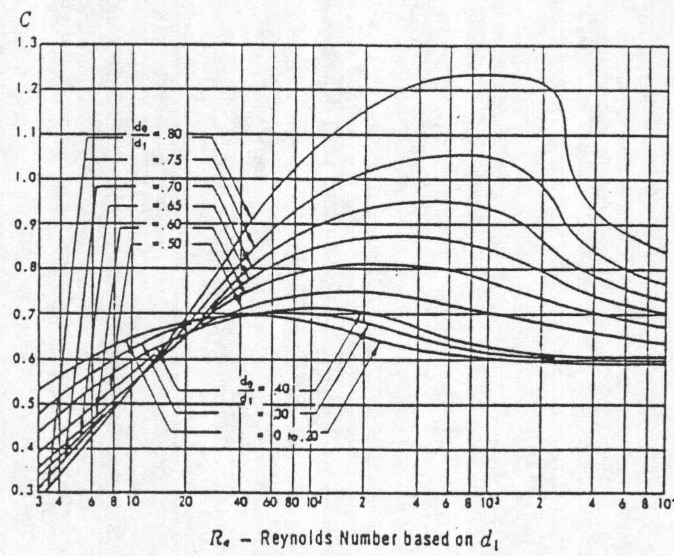
Example: The flow coefficient C for a diameter ratio d_0/d_1 of 0.60 at a Reynolds number of 20,000 (2×10^4) equals 1.01.

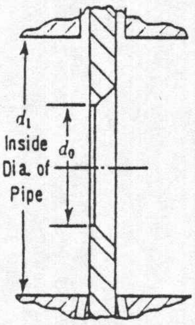


Flow Coefficient C for Square Edged Orifices[7, 17]

Orifice Plate

$$C = \frac{C_d}{\sqrt{1 - \left(\frac{d_0}{d_1}\right)^4}}$$





Flow →

$$C = \frac{C_d}{\sqrt{1 - \left(\frac{d_0}{d_1}\right)^4}}$$

Lower chart data from *Regeln fuer die Durchflussmessung mit genormtem Duesen und Blenden*. VDI-Verlag G. m.b.H., Berlin, SNW, 7, 1937. Published as Technical Memorandum 952 by the NACA.